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MODULAR TROUGH POWER PLANTS

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ABSTRACT

A number of factors are creating an increased market potential for small trough power technology. These include the need for distributed power systems for rural communities worldwide, the need to generate more electricity by non-combustion renewable processes, the need for sustainable power for economic growth in developing countries, and the deregulation and privatization of the electrical generation sector worldwide. Parabolic trough collector technology has been used in large central station power plants. Organic Rankine cycle (ORC) air-cooled modular power units have been successfully applied for large and small-scale geothermal power plants, with over 600 MW of capacity, during the same period. The merging of these two technologies to produce distributed modular power plants in the 200 kW to 10 MW range offers a new application for both technologies. It is our objective in this paper to introduce a modular trough power plant (MTPP) and discuss its performance and the cost of electricity generation from such system.

INTRODUCTION

Parabolic trough technology has proven to be a very mature solar technology for large-scale power generation. The nine Solar Electric Generating Systems (SEGS) parabolic trough power plants in the California Mojave Desert consists of 354 Megawatts of installed electric generating capacity that have been in operation for up to 16 years. The SEGS plants utilize steam Rankine cycle power plants. Economic optimization of steam power systems for bulk power applications tends to drive plants to larger and larger sizes. Unfortunately, low energy prices in recent years have slowed the continued development of large-scale trough power plants for bulk power markets. Higher value market opportunities for trough solar power plants include smaller distributed generation and remote power applications. Distributed generation has higher value than centralized power generation because it can eliminate power losses in the transmission and distribution (T&D) system, improve system reliability, and occasionally offset the need for upgrades to the T&D system. Distributed generation located at a customer site often offsets energy costs at the customer's retail price rather than the utility's price for bulk generation. Remote power applications are typically high value because of high fuel prices and low conversion efficiencies. Diesel generators or photovoltaics are often the competition for remote power applications. In addition, a number of green power markets are developing where customers either choose or are obligated to purchase renewable electric power. Unfortunately, the green power market has not matured to the point where it will support the development of large solar power plants. This paper looks at the opportunity for developing smaller trough power plants that might be suitable for distributed, remote, or green power markets. Because of the inherent problems (complexity and operational issues) with steam cycle power plants at smaller sizes, this paper focuses on systems that integrate troughs into organic Rankine cycle (ORC) power plants.

In the early 1980's, the Coolidge Solar Irrigation project (1983) demonstrated a 150 kWe trough ORC solar power plant. This plant operated successfully for several years but suffered from a number of problems, that at the time precluded further development of this concept. The main problems were low collector performance, high operation and maintenance (O&M) costs mainly due to the problems associated with the cooling tower, and a low annual output.

Given the significant improvements in solar and ORC technologies since the late 1970's, a reassessment of the technology is warranted.

ORGANIC RANKINE CYCLES (ORC)

ORC power cycles are primarily used for lower temperature heat sources such as geothermal or waste heat recovery. The low resource temperature results in low efficiency of the ORCs, however, ORCs can be designed to operate at substantially higher efficiencies for trough systems. Hundreds of megawatts of ORC power systems have been installed around the world. ORCs use organic (hydrocarbon) fluids that can be selected to best match the heat source and heat sink temperatures. They can use air-cooling instead of evaporative wet cooling typically used at steam Rankine cycle plants. The hydrocarbon working fluids work just like steam does in the steam Rankine cycles, however, the ORC fluids are generally used at lower pressures and for safety reasons are condensed at above atmospheric pressures. These factors greatly reduce the complexity and cost of ORC systems. In addition, smaller ORC systems can generally be run remotely and only periodically need on-site operator or maintenance intervention.

The primary advantages of an ORC power cycle for applications with troughs are:

- ORCs operate at lower temperatures and thus we can reduce trough operating temperatures from 735F (390 C) to 580F (304 C). This means that an inexpensive heat transfer fluid such as Caloria may be used instead of the existing fluid. Since Caloria is inexpensive, it can be used in a simple 2-tank thermal storage system similar to the thermal storage system at SEGS I.
- Lower solar field operating temperatures are likely to translate into lower capital cost and more efficient solar field equipment.
- ORCs can be designed to use air-cooling for the power cycle. This and the fact that the power cycle uses a hydrocarbon for a working fluid (instead of steam) means that the plant needs virtually no water to operate. Mirror washing is about 1.5% of the water use at the SEGS. This means that the plants can be built in desert locations that have limited water availability.
- ORC power cycles are simple and generally can be operated remotely. This helps to reduce O&M costs which has been one of the key reasons for concentrated solar power (CSP) technologies to increase in size.
- Economies-of-scale can be improved through the development of standardized designs and modular systems. Solar technology has the advantage that the solar field can be sized and designed to meet the requirements of the local solar resource while the power plant design remains unchanged. This reduces the initial design cost and allows for mass production of the power cycle components, specifically the turbine.

ORC systems have a number of disadvantages as well. First, ORC systems generally have lower efficiencies than steam cycles that run at higher temperatures and pressures. However, the efficient steam cycles (approximately at 35% net) come at the price of more capital investment and the need for higher resource temperatures. The use of air-cooling means that ORC cycles are negatively impacted by high ambient temperatures. However, many high desert locations have good solar resources and cooler ambient conditions.

TROUGH ORC SYSTEM

Given the potential advantages that an ORC power plant could offer, an investigation was undertaken to evaluate the potential for a 1 MWe trough power system based on current solar and ORC technologies. General design constraints include the use of dry cooling which has been applied to geothermal power plants over the past 20 years, and the use of Caloria heat transfer fluid to allow the integration of thermal storage for power generation during periods with no or low solar radiation. The modular nature of these systems simplifies siting requirements due to smaller size, minimizes on-site erection through skid mounting, and provides the possibility of prepackaging collector and power cycle hardware and shipping materials to site in containers. It also minimizes O&M costs due to increased use of self-diagnostics. The goal of the trough ORC power system will be to create an automated and virtually unattended trough power plant. This concept blends two field proven technologies into a new solar power system with potential markets in the USA for distributed power, off grid or grid connected, and for rural electrification applications in developing countries.

MTPP ORGANIC RANKINE CYCLE ANALYSIS

Organic Rankine power cycles are typically used for applications with low resource temperatures such as waste heat recovery or geothermal applications. In these cases, the objective is to get the most specific power possible from a particular thermal resource while preventing the resource from depleting. These are in essence once through systems where you use the energy or lose it. However, in solar applications the goal is to develop the most efficient solar and power plant systems while trading off the capital and O&M cost of various components. Since ORC power cycle efficiency tends to be a stronger function of temperature than trough solar fields the optimum trough ORC system will likely have a high average solar resource temperature.

The ORC analysis presented here utilizes a solar resource temperature of 580F (304 C). This corresponds to the reasonable safe upper operating limit of Caloria. Using this as a boundary condition, an analysis of potential ORC configurations was conducted using the commercially available ASPEN thermal process modeling software (Aspen Technology, 2000). Three ORC cycles were analyzed in this work: a simple Rankine cycle, a Rankine cycle with recuperation, see Figure 1, and a simple Rankine cycle with reheat and recuperation, see Figure 4. Pentane was used as the working fluid for all these cycles because it provided the best match for the resource temperature while allowing above atmospheric pressure in the condenser. These cycles and their corresponding performance information are listed in Table 1.

TABLE 1: ORCS ANALYZED FOR 580 F (304 C) RESOURCE TEMPERATURE

Cycle	Cold HTF ºF (C)	HTF Flow lb/hr (kg/s)	Gross Elec. kW	Parasitic Elec. kW	Net Effic. %
Basic Rankine	181 (82.7)	92,264 (11.65)	1093	122.1	12.5
Recuperated	344 (173.3)	91,771 (11.59)	1093	124.0	20.1
Recup. & Reheat	415 (212.8)	131,644 (16.62)	1125	123.3	20.5

All Cycles

1. Solar Resource (Hot HTF) temperature 580ºF (304 C)
2. Boiler pressure 640 psia (4.4 MPa)
3. Condenser pressure 20 psia (0.138 MPa)

The main assumptions used for our analysis were: resource temperature = 580 F (304 C), sink temperature = 80 F (26.7 C), turbine efficiency of 0.75, pump efficiency of 0.67, generator efficiency 0.94, and recuperator effectiveness of 0.80. A pinch point of 17 F (9.4 C) was assumed for the heater/boiler while the pinch point for the air cooler/condenser was assumed to be 13 F (7.2 C). Table 1 shows that the efficiency of the basic Rankine cycle is very low, only 12.5%. The addition of recuperation significantly improves the overall efficiency of the cycle. The cycle with reheat and recuperation has a slightly higher efficiency than that of the Rankine cycle with recuperation only. Considering the complexity of the cycle when reheat is used, and considering that reheat does not add much to the cycle efficiency, we decided to focus on the cycle with recuperation only.

Recuperated Organic Rankine Cycle: The cycle of interest is shown in Figure 1. In this analysis, an air-cooled condenser was used and the sink temperature was assumed to be 80 F (26.7 C). The resource entered the heat exchanger at 580 F (304 C) and its exit temperature was 344 F (173.3 C). The working fluid, Pentane, was pumped from a pressure of 20 psia (0.138 MPa) at 114 F (45.6 C)(saturated liquid conditions) to a pressure of 650 psia (4.48 MPa) where it was heated to a temperature of 325 F (162.8 C) inside a recuperator by the stream exiting the turbine. The pressure drop inside the recuperator was assumed to be 10 psia (69 kPa). Then it was passed through the main heat exchanger where it was boiled and heated by the oil from solar field to a temperature of 563 F (295 C). The vapor exiting the boiler was then passed through a turbine and was allowed to expand to 24 psia (0.165 MPa). The stream exiting the turbine was passed through a recuperator to heat the feed working fluid. The stream exiting the recuperator was sent through an air-cooled condenser where the working fluid was completely condensed. The efficiency of this cycle

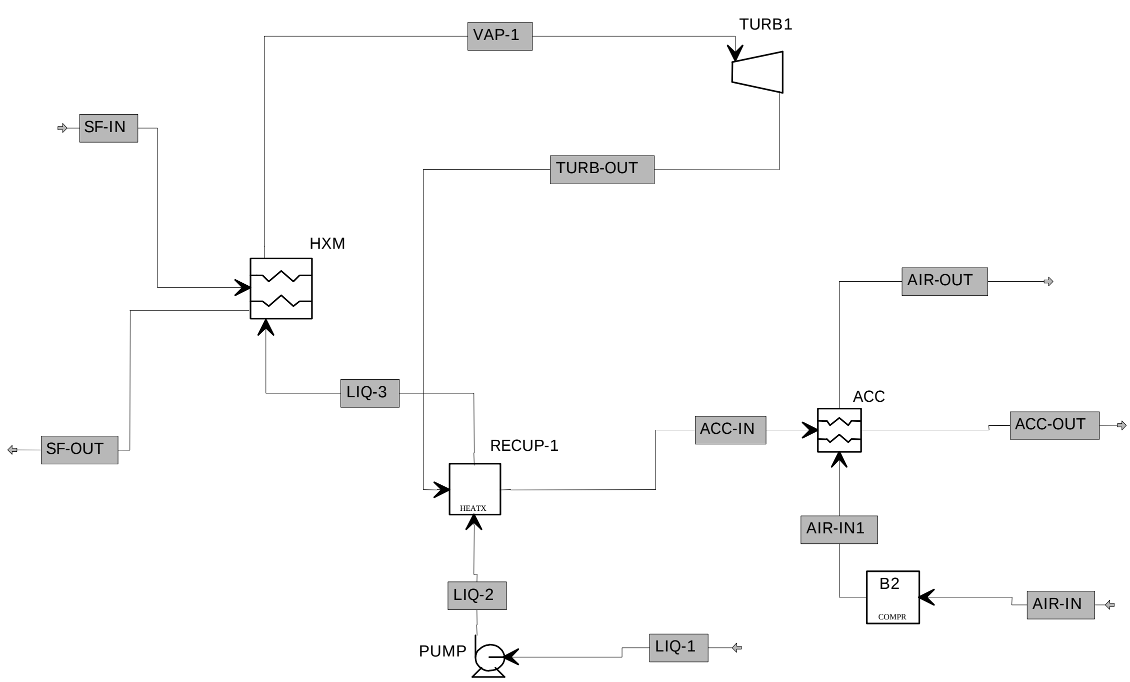

Figure 1: Organic Rankine Cycle with Recuperator

including the fan power for the air cooler and the pump power is 20.1%. A pressure drop of 20 psia (0.138 MPa) has been assumed for all runs through the boiler/superheater. A total pressure drop of 4 psia (27.6 kPa) was assumed for the hot side of the recuperator and the air-cooled condenser. The heating and cooling curves for this cycle have been presented in Figures 2 and 3. Figure 2 shows that by choosing a supercritical pressure of 640 psia (4.4 MPa), the heating curve of the working fluid matches the cooling curve of the solar heat transfer fluid very closely reducing the irreversibilities that occur in the boiler. The cooling curve of the working fluid, as shown in Figure 3, shows that the condensation occurs at a constant temperature (except for the pressure drop effect in the piping). The pinch point of 13 F (7.2 C) occurs at the start of condensation.

Note that a conservative condensing pressure of 20 psia (0.138 MPa) was chosen for this analysis. It is possible to condense this working fluid at pressures as low as 15 psia (0.103 MPa), however, an above atmospheric condensing pressure is very desirable for this cycle. It is also necessary to optimize the condensing pressure with respect to the cycle efficiency. This type of optimization is out of the scope of this work, however, a very preliminary attempt indicates that at a condensing pressure of about 17 psia (0.117 MPa), an efficiency of 20.3% can be achieved.

It is possible to increase the effective area of the recuperator to enhance the performance of the cycle, however, proper precautions should be taken to account for the pressure drop and the heat transfer coefficient when the area is increased.

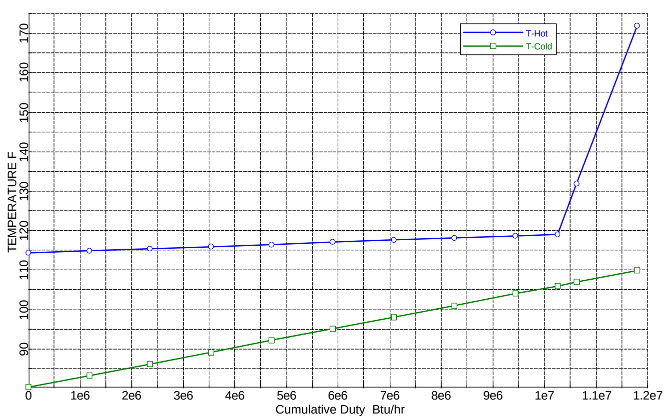

Figure 2: Boiler Heating Curve for Cycle in Figure 1

<u>Recuperated Organic Rankine Cycle with Reheat</u>: The other cycle of interest that was analyzed was a Rankine cycle with both reheat and recuperation as shown in Figure 4. In this analysis, an air-cooled condenser was used and the sink temperature was assumed to be 80 F (26.7 C). The resource entered the heat exchanger at 580 F (304 C) and its exit temperature was 415 F (212.8 C).

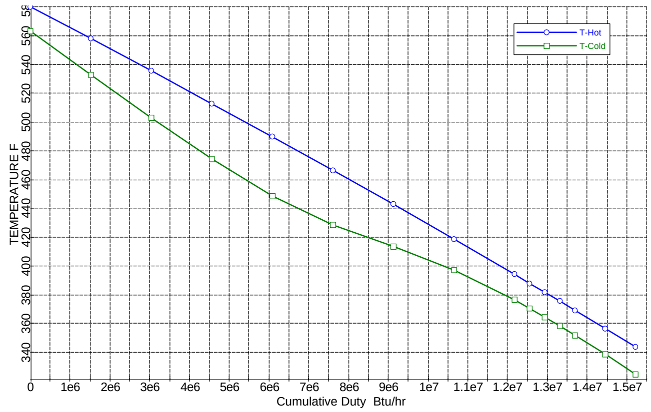

Figure 3: Condenser Cooling curve for Cycle in Figure 1

The working fluid, Pentane, was pumped from a pressure of 20 psia (0.138 MPa) at 114 F (45.6 C) (saturated liquid conditions) to a pressure of 650 psia (4.48 MPa) where it was heated to a temperature of 348 F (175.6 C) inside a recuperator by the stream exiting the low pressure turbine. Again a 10 psia (69 kPa) pressure drop was assumed for the recuperator. Then it was passed through the main heat exchanger where it was boiled and heated by the oil from solar field to a temperature of 563 F (295 C). The vapor exiting the boiler was then passed through a high-pressure turbine and was allowed to expand to 330 psia (2.27 MPa) where it was superheated to a temperature of 563 F (295 C). This vapor was passed through a low-pressure turbine and was expanded to a pressure of 24 psia (0.165 MPa). The exhaust of the low-pressure turbine was passed through a recuperator to heat the feed working fluid. The stream exiting the recuperator was sent through an air-cooled condenser where the working fluid was completely condensed. A pressure drop of 20 psia (0.138 MPa) has been assumed for all runs through the boiler/superheater and a total pressure drop of 4 psia (27.6 kPa) was assumed for the recuperator and the air cooled condenser. The efficiency of this cycle including the fan power for the air cooler and the pump power is 20.5%.

<u>Options for Improving Cycle Efficiency</u>: The above cycles were analyzed for a condensing pressure of 20 psia (0.138 MPa), and a pinch point of 17 F (9.4 C) for the boiler and 13 F (7.2 C) for the condenser. To identify the highest efficiency that one could get for the cycle with recuperator, we decided to lower the condensing pressure to 17.5 psia (0.121 MPa), reduce the pinch point in all the heat exchangers to 5 F (2.8 C) while maintaining the turbine efficiency at 0.75. In doing so, we obtained a net cycle efficiency of 23.0%. This is a considerable improvement over the 20.1% efficiency that we were getting earlier. However, the specifications for this cycle need to be further verified and validated.

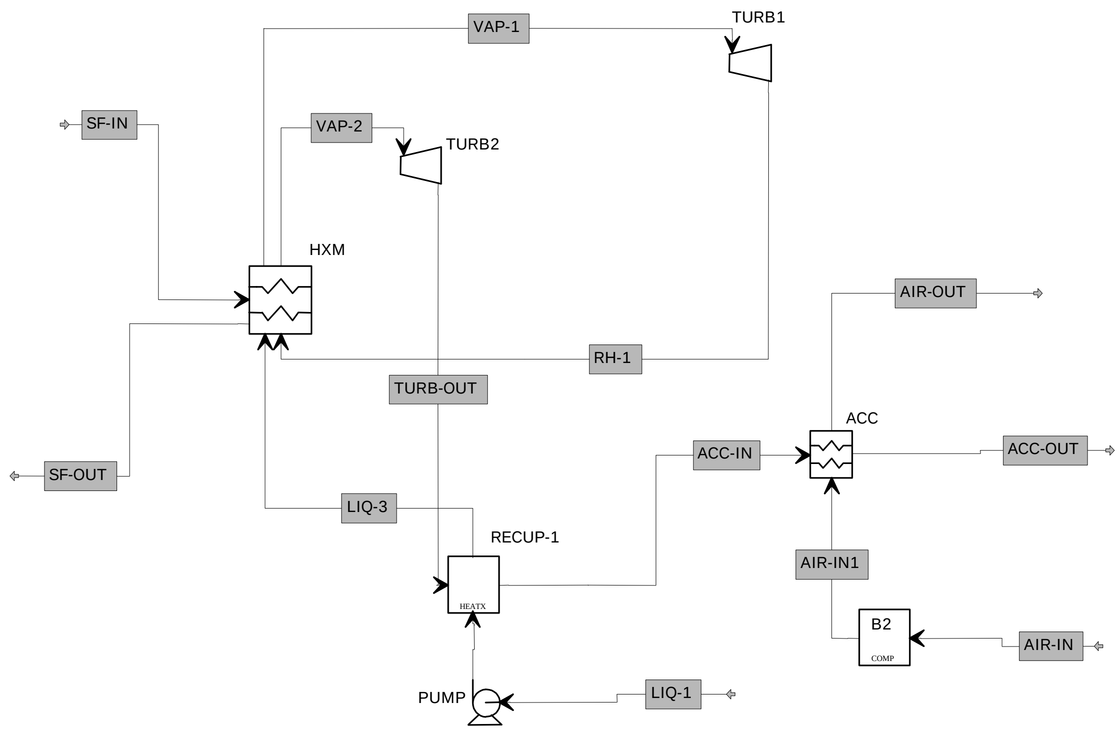

Figure 4: Organic Rankine Cycle with Recuperator and Reheat

Organic Rankine Cycle Cost for MTPP: The installed capital cost for the power cycle side of a MTPP is estimated to be about $1700 per kW based on a preliminary analysis carried out by Barber-Nichols (2000). The major costs are associated with the turbine and the air-cooled condenser. The air-cooled condenser constitutes about 20% of the total capital cost while turbine cost is about 45% of the total cost. The recuperator which improves the cycle performance significantly requires only 6% of the total cost while the boiler requires about 15%. The remaining 14% of the capital cost is spent on installation, pump, and other miscellaneous components. By creating modular designs, the goal is to develop packaged systems with an installed capital cost of under $1000/kWh.

SOLAR TECHNOLOGY

In general, two parabolic trough solar collector technologies have been considered for integration into the ORC plant. The first is the Luz second-generation collector known as
the Luz System Two (LS-2). The second collector is the Industrial Solar Technology (IST) parabolic trough solar collector. Table 2 highlights the key design parameters of these two collectors. Both collectors have extensive field operational experience and have been tested at the National Solar Thermal Test Facility at Sandia National Laboratories in Albuquerque, New Mexico (Dudley, 1994 and 1995). Figure 5 shows the thermal performance of both collectors over a range of operating temperatures.

The LS-2 collector utilizes a torque tube galvanized steel structure with a silvered glass reflector and an evacuated receiver with a Cermet selective coating. The LS-2 collector has demonstrated excellent performance, high availability, and ease of installation and maintenance. There are several concerns with the LS-2 collector: the capital cost, the lifetime of it's evacuated receiver, and that no company currently markets the LS-2 design. Recent cost studies by Flabeg Solar International (Pilkington, 2000) have estimated collector costs around $200 per square meter for the third generation Luz parabolic trough collector design (LS-3) for a solar field of 200,000 to 300,000 square meters. Given roughly similar components and weight, it is assumed that the LS-2 collector is approximately similar in cost. Much effort is currently being focused on resolving the receiver reliability issues, which seems to be related to reliability of a glass to metal seal that is necessary for an evacuated receiver. One option under consideration for ORC plants is to use a non-evacuated receiver. Although this will result in reduced solar field performance, it could also reduce cost and improve collector field reliability. Figure 5 indicates a drop of less than 5% by going to a non-evacuated receiver. Also note that the solar collector efficiency at ORC temperatures is about the same with a non-evacuated receiver as it is for an evacuated receiver operating at the temperatures required for a steam power plant. Collectors for smaller ORC plants would likely be higher in cost, so some focus is required to find ways to reduce installation and the transaction costs for small systems. This collector would be considered only if it is also being considered for use with one or more of the large trough projects currently under development.

TABLE 2: COLLECTOR TECHNOLOGY

	Luz LS-2	**IST**
Concentrator		
Aperture	5 m	2.3 m
Length	47 m	6.1 m
Aperture Area	235 m^2	
Focal Length	1.84 m	0.762 m
Rim Angle	70 deg	72 deg
Concentration Ratio	71	45
(Mirror Aperture to Receiver Diameter)		
Mirror Reflectivity	0.93 (est.)	0.93
Optical Efficiency	0.733	0.778
Receiver		
Operational Range	100-400ºC	100-300ºC
Tube Diameter (I/O)	70 mm	51 mm
Length	4 m	6.1 m
Glass Envelope Transmittance	0.96	0.96
Selective Coating	Cermet	Black Nickel
Absorptance	0.96	0.97
Emmitance	0.14	0.30
	@ 350C	@ 300C
Collector Efficiency		
Non-evacuated @ 215C	66.1%	61.9%
Evacuated @ 215C	69.4%	

The IST collector utilizes an aluminum structure and uses a silvered or aluminumized polymeric reflector bonded to aluminum sheets. The receiver is a non-evacuated design that uses a black nickel selective coating with a solgel antireflective coating on the glass envelop. The IST collector has been used primarily for lower temperature process heat applications in field sizes of approximately 2000 to 5000 square meters. The IST collector has been tested at Sandia National Laboratory at temperatures up to 350 C (Dudley, 1995). The IST collector field operates unattended and requires minimal on-going maintenance. The primary concerns with the IST collector are the lifetime of the silverized polymeric reflector. The installed cost of the IST collector is about $200 per square meter. IST is currently looking at alternative reflectors, conversion of the structure to steel, and the potential to increase the collector size.

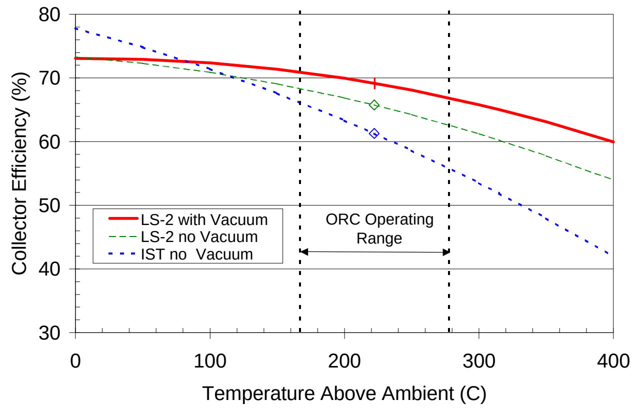

Figure 5: Sandia Parabolic Trough Collector Test Results (Dudley 1994).

The Sandia test data in figure 5 indicates the IST collector performs about 6% below the non-evacuated LS-2 collector at the average temperature needed for the ORC plant. The lower performance is due primarily due to the lower concentration ratio of the IST collector and the high emittance of the Black Nickel selective coating.

THERMAL STORAGE SYSTEM

The ORC solar resource temperature has been defined by the selection of Caloria as the heat transfer fluid to be used in the analysis. This fluid is a low-cost mineral oil that can be used as a cost effective form of thermal storage. The same fluid was used at the SEGS I project in a 2-tank thermal storage system. The SEGS I storage system had 3 hours of themal storage capacity and operated on a daily basis for dispatching solar electric generation to the utility's high energy demand period for 13 years. This system was destroyed by a fire in 1999 after an apparent double failure of system used to maintain a positive pressure in the storage tanks. Given the 13 years of successful operation, it seems likely that a design fix could resolve the concerns with this system. However, special care will be necessary to minimize the potential fire risk.

SYSTEM ANALYSIS

A preliminary analysis has been completed to assess the potential economic feasibility of small trough ORC power plants. NREL has developed a hourly simulation model capable of modeling the performance of parabolic trough solar power plants. This model has been validated against the actual steam Rankine cycle parabolic trough power plants and found to reproduce annual electric performance within a few percent. Using the ORC power cycle performance for the system developed by Barber Nichols, NREL has modified the trough power plant model predict the performance from a parabolic trough ORC plant. A nominally 1MWe net parabolic trough ORC power plant with thermal storage was modeled for this analysis. Table 3 highlights the key plant design assumptions.

TABLE 3 MTPP DESIGN ASSUMPTIONS & PERFORMANCE

Location
- Barstow, California
- Annual Direct Normal Radiation 2800 $kWh_t/m2$

Power Cycle
- 1 MWe (net electric generation)
- Recuperated Organic Rankine Cycle
- Air Cooling - 80 F (27C) design point
- 22.5% thermal to net electric efficiency
- Capital Cost; \$1700/kW_e (Barber Nichols, 2000)

Collector Field
- Luz LS-2 Collector
- Collector field temperature 380-580F (193-304C)
- Receiver: Cermet selective coating, non-evacuated receiver
- Collector Cleanliness 90%
- Solar Field Availability 99%
- Size: 20,000 m^2 @ \$200/m^2

Thermal Storage
- Heat transfer fluid: Caloria HT-43
- 2-Tank Thermal storage system
- 9 hours of thermal storage (47 MWh_t)
- Cost: \$10/kWht

Plant Performance (modeled)
- Annual Solar Field Efficiency 44%
- Annual Heat Losses from Storage 2.2%
- Dumped Energy (storage full) 1.3%
- Annual Net Electric Output 4632 MWh_e
- Capacity Factor @ 1 MWe: 53%
- Annual Solar to Electric Efficiency: 8.4%

Plant Cost
- 10% Engr. Design, Construction Mgt., Contingency
- Capital Cost, \$7044/kWe

Economic Assumptions
- Lifetime, 20 years
- Discount Rate, 10%
- Annual Insurance, 0.5% of Capital Cost

O&M cost
- Solar Field, 1.0¢/kWh
- Power Plant Maintenance/Operation, 1.5¢/kWh

Levelized Energy Cost, 21¢/kWh

The parabolic trough performance model calculates the solar field thermal delivery on an hourly basis. The solar energy is stored in the thermal storage system. The model has a dispatch strategy to determine when the power plant should be operated. Solar energy can be dispatched for night time electric production. The model includes thermal losses from the solar plant and thermal storage and calculates the parasitic electric consumption of the plant for both online and offline. The annual performance of the trough ORC plant is shown in table 3.

A simple levelized energy cost (LEC) calculation is used for the economic assessment. Table 3 shows the assumptions used for the analysis. A 20-year lifetime is assumed. The operation and maintenance costs are low for this plant because the power plant is assumed to operate automatically without the need for an on-site operator. The power plant O&M costs are consistent with small remotely operated geothermal power plants. The solar field costs are in line with other small trough plants. Based on these assumptions, the Levelized energy cost is about 21¢/kWh for this plant. Assuming a 90% learning curve (a 10% reduction in cost with every doubling of installed capacity). future costs are expected to drop to below 15 ¢/kWh after a few tens of systems are installed.

CONCLUSIONS

ORC power cycles and parabolic trough solar collector technology have been successfully demonstrated separately. With the current state of these technologies, the modular trough power plant is a technologically viable concept. Our analysis indicates that cycle efficiencies in the range of 23% for a solar resource temperature of 580 F (304 C) are possible. Integrating this with current parabolic trough collector and thermal storage technologies allows for solar power plants that can dispatch solar power efficiently at any time of day. A cost of power around 20¢/kWh appears to be feasible with current technology. These costs look to be attractive for remote power needs in sunny regions of the developing world where low labor rates could further reduce the cost of power.

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